Fermilab

Particle Physics Division
Mechanical Department Engineering Note

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Project Internal Reference:

Project: **ILC**

Title: Vacuum Vessel Engineering Note for 3.9 GHz, 3rd Harmonic

Author(s): Edward Chi

Reviewer(s):

Key Words:
Crymodule, Vacuum, Pressure, FEA, Vessel Shell, Stiffening Rib, Reinforcement, Flange, Weldment, Up & Downstream,

Abstract Summary:
The vacuum vessel of 3.9 GHz, 3rd harmonic has 42.87" O. D, ~ 82.64" Length with multiple openings for different connections and instrumentation. This engineering note is presented extensive discussion, FEA analysis and calculations for the vessel and nozzle shells, stiffening rings, multiple openings and reinforcements, different flanges, welds and others per Fermilab, ASME and other applicable codes.

Applicable Codes:
ES&H Manual Chapter 5033, Fermilab.
"Boiler & Pressure Vessel Code" ASME VIII, Div.1
Addenda July 1, 2003
"Allowable Stress Design", AISC, 9th edition
"Structural Welding Code-Steel", AWS D1.1-90

EXHIBIT A-1

Vacuum Vessel Engineering Note
(per Fermilab ES&H Manual Chapter 5033)

Prepared by	Edward Chi	Date	08-26-2005	Div/Sec	PPD/MD/ME
Reviewed by		Date		Div/Sec	
Div/Sec Head		Date		Div/Sec	

1. Identification and Verification of Compliance

Fill in the Fermilab Engineering Conformance Label information below:

This vessel conforms to Fermilab ES&H Manual Chapter 5033

Vessel Title	*3.9 GHz, 3rd Harmonic Vacuum Vessel*
Vessel Number	
Vessel Drawing Number	*5520 – ME - 439317*
Internal MAWP	*1 psig*
External MAWP	*14.7 psi*
Working Temperature Range	*20* °F *100* °F
Design/Manufacturer	*DESY*
Date of Manufacture	
Acceptance Date	

Director's signature (or designee) if vessel is for manned area and requires an exception to the provisions of this chapter.

____

Amendment No.	**Reviewed by:**	**Date:**

Laboratory location code ____________________
Laboratory property number ____________________
Purpose of vessel ____________________

List all pertinent drawings

Drawing No.	Location of Original
5520 – ME – 439317 -1	
5520 – ME – 439317 -2	
5520 – ME - 439262	

2. Design Verification

Provide design calculations in the Note Appendix.
See the attached vacuum vessel engineering note #MD-Eng-088

3. System Venting Verification

Can this vessel be pressurized either internally or externally? [ ] Yes [X] No

If "Yes", to what pressure? ____________

List all reliefs and settings. Provide a schematic of the relief system components and appropriate calculations or test results to prove that the vessel will not be subjected to pressures greater than 110% beyond the maximum allowable internal or external pressure.

Manufacturer	**Relief**	**Pressure Setting**	**Flow Rate**	**Size**

4. Operating Procedure Section

Is an operating procedure necessary for the safe operation of this vessel?
[ ] Yes [X] No *(If "Yes", it must be appended)*

Is a testing procedure necessary for the safe acceptance testing (acceptance testing) of this vessel?

[ ] Yes [X] No

If "Yes", the written procedure must be approved by the division head prior to testing and supplied with this Engineering Note.

5. Welding Information

Has the vessel been fabricated in a Fermilab shop? [ ] Yes [X] No

If "Yes," append a copy of the welding shop statement of welder qualification.

The fabrication will be performed by using qualified welder of an outside contractor.

6. Exceptional, Existing, Used and Non-Manned Area Vessels

Is this vessel or any part thereof in the above categories? [ ] Yes

[X] No

If "Yes" follow the Engineering Note requirements for documentation and append to note.

CONTENTS

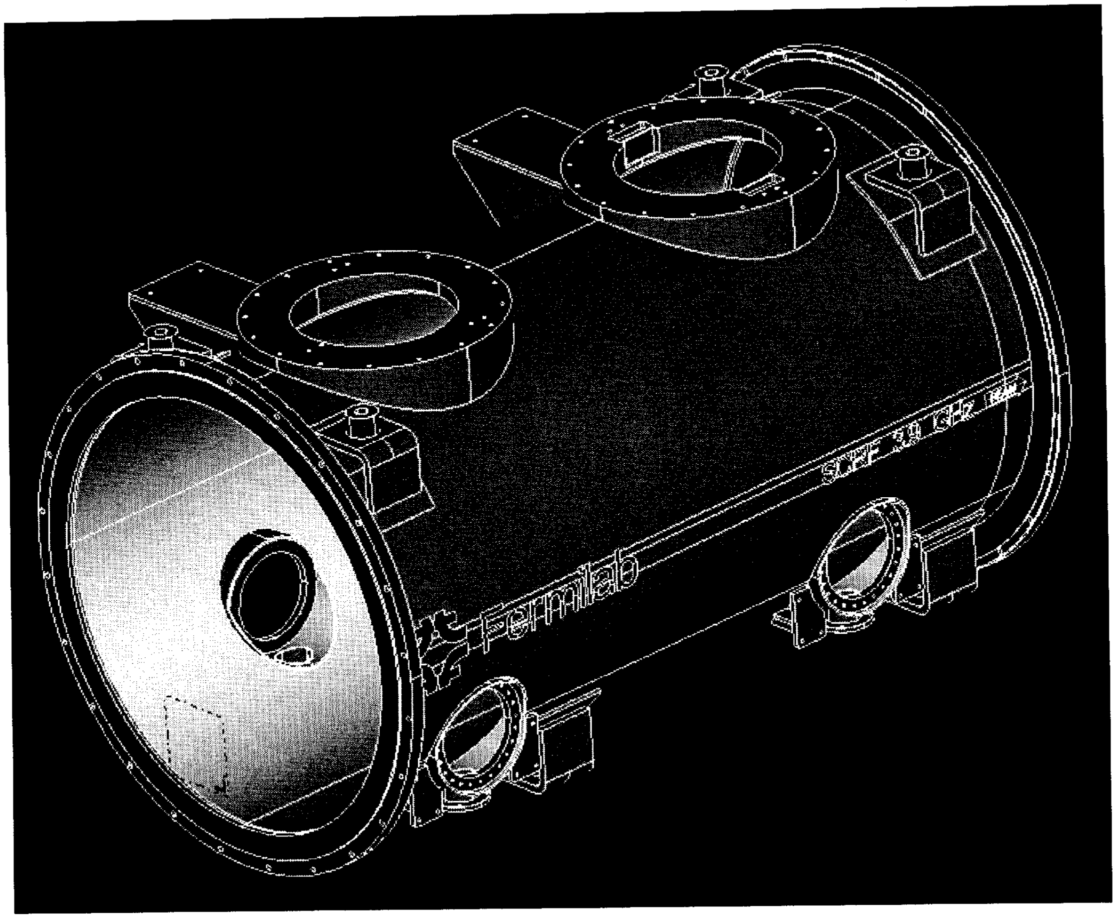

Figure i.1, An overall view of the vacuum vessel weldment of 3.9 MHz, 3rd Harmonic

Reference drawing: ME – 439317- 1(2)
Overall length = 82.64 in
$W_{ws} \approx 2{,}264$ lbs. the weight of the vessel shell weldment.

Assuming the vessel shell is operating under the room temperature.
Vessel shell material: SA 516-60
Materials for flanges, ports and nozzles: see respective notes and drawings.

1. Calculation for the Vacuum Vessel Cylindrical Shell Thickness

The Vacuum Vessel Cylindrical Shell:

Given:

P: 14.7 psi, the external design pressure,
D_o: 42.87 in, outside diameter of the vessel shell
D_i: 42.24 in, inside diameter of the vessel shell
t_n: The nominal vessel shell thickness.
F_a: 15,000 psi, the max. allowable stress of the shell material at the operating temp, also know as S_V.
Vessel shell material: SA 516-60, carbon steel, see dwg. #ME-439262
L_v: Design length of the vessel between lines of the support.
M.A: Material corrosion and other misc. allowance, ~ 1/16"

Assuming:
$L_v \approx 82.64$ in (per drawing #ME-439317-01)

Find out the required vessel shell thickness t_n:

1. Try $t_n = 0.31"$
   $L_v/D_o = 82.64" / 42.87" = 1.93$
   $D_o/t_n = 42.87" / 0.31" = 138.29$

   Per section UG-28, also Fig. G and Fig. CS-2 in Subpart 3 of Section II, part D, ASME VIII, Div. 1:
   It is found out that:
   $A = 0.00043$
   $B = 6000$
   Then:
   $P_a = (4B) / (3D_o/t_n)$
   $= (4 \times 6000) \div (3 \times 138.29)$
   $= 57.85$ psi
   where:
   P_a : The max. allowable external working pressure, psi
   A: Factor determined from Fig. G
   B: Factor determined from the applicable material chert in Fig. CS-2.

2. Try $t_{r2} = 0.2$
   $L_v/D_o = 1.93$
   $D_o/t_{r2} = 42.87"/0.20" = 214$

   Find:
   $A \approx 0.00023$
   $B = 3300$
   Then:

$P_{a2} = (4B) / (3D_o/t_n)$
$= (4 \times 3300) \div (3 \times 214)$
≈ 19.64 psi

3. Try $t_{r3} = 0.175$
   $L_v/D_o = 1.93$
   $D_o/t_{r3} = 42.87"/0.175 = 245$

   Find:
   $A \approx 0.00019$
   $B = 2800$

   Then:
   $P_{a3} = (4B) / (3D_o/t_n)$
   $= (4 \times 2800) \div (3 \times 245)$
   ≈ 15.23 psi

   So, **t_r = 0.175"**

So the vessel shell thickness is required 0.175", using 0.31" as vessel shell thickness is above the shell thickness required and it is satisfactory.

2. The Permissible Out of Roundness of the Vessel Cylindrical Shell

Given:

P: 14.7 psi, the external design pressure,
D_o: 42.87 in, outside diameter of the vessel shell
D_i: 42.24 in, inside diameter of the vessel shell
t_n: 0.31", the nominal vessel shell thickness.
F_a: 15,000 psi, the max. allowable stress of the shell material at the operating temp, also know as S_v.
Vessel shell material: SA 516-60, carbon steel, see dwg. #ME-439317
L_v: ~82.64", design length of the vessel between lines of the support.

The difference between the max. dia and the mini. dia. of the vessel shell at any cross section shall not exceed:

$0.01 \times 42.87 \text{ in} = 0.428 \text{ in}$
(per section UG-80 (b)(1) of ASME VIII, Div.1)

Per section UG-80(b)(2) and figure UG-80 of ASME VIII, Div.1, it is found out that the maximum plus or minus deviation from the true circular form, measured radially on the outside or inside of the cylindrical vessel shall not exceed the maximum permissible deviation e:

$$e \approx 0.65t$$
$$= 0.65 \times 0.31 \text{ in}$$
$$= 0.202 \text{ in}$$

where:
$D_o / t = 138$
$L_v / D_o = 1.93$

Per Figure UG-29.2, section UG-29 of ASME VIII, Div.1, it is found out that:

$$\text{Arc length} \approx 0.20\, D_o$$
$$= 0.20 \times 42.87 \text{ in}$$
$$= 8.574 \text{ in}$$
$$\text{Chord length} = 2 \times \text{Arc length}$$
$$= 2 \times 8.574 \text{ in}$$
$$= 17.15 \text{ in}$$

So, in a chord length of 17.15 in, the maximum plus or minus deviation from the true circle form shall not exceed 0.202 in.

3. Analysis and Calculation for the Stffening Rings

3.1 To Define the Downstream End Flange as the Stiffening Ring for the Vessel Shell Under the External Pressure

Reference drawings: ME-439317, MD-439240

Givien:

$E = 28 \times 10^6$ psi, Modules of elasticity of the flange materials

$l_e = 1.1\ (D_0 t)^{0.5} = 1.1\ (42.87" \times 0.31")^{0.5}$

$= 4.01"$, the effective regional length of the vessel shell

F_y: 25 ksi, min. yield stress of the stiffening material A182-F304L under the operating temperature.

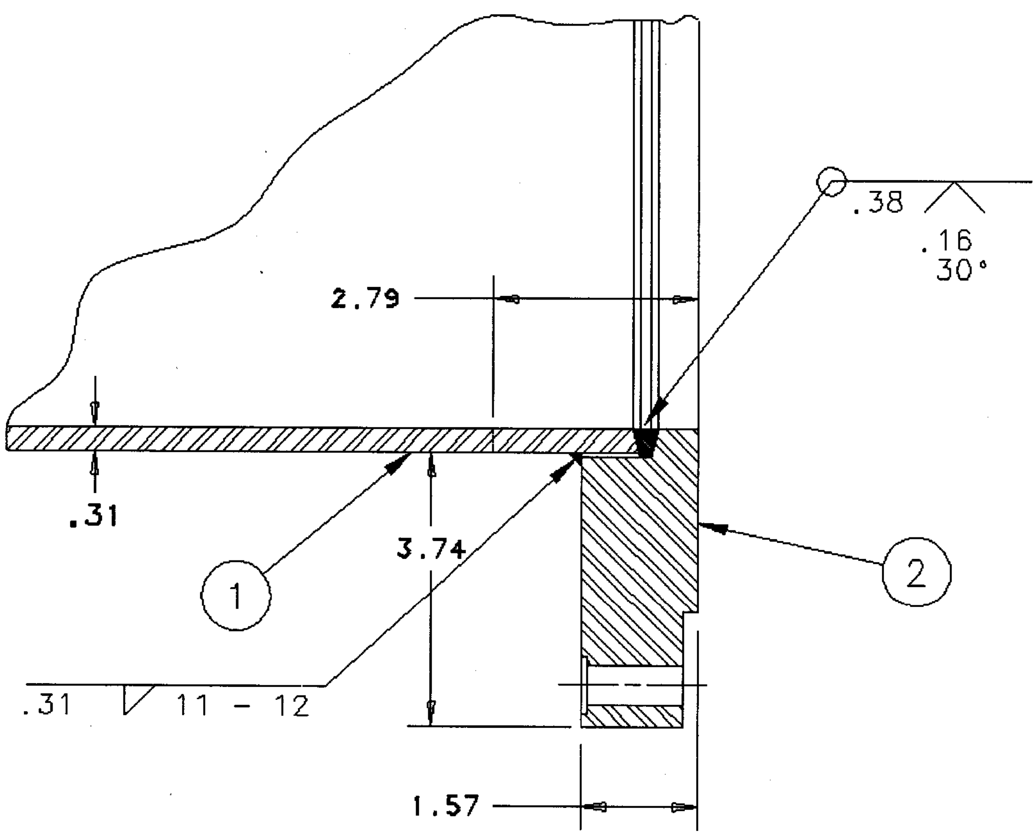

Figure 3.2, The cross-section view of the end flange with the vessel shell, also refer to detail L of drawing ME-439317, and drawing MD-439240..

To find the required moment of inertia of the combined end flange – shell cross section about the neutral axis parallel to the axia of the shell, in^4, I_s':

$I_s' = [D_o^2 L_s(t + A_s/L_s)A] / 10.9$ (eq. 3-2)
(per UG-29, ASME VIII, Div.1)

Where:

A_{sd}: Cross-sectional area of the downstream end flange stiffening area, in^2
$\approx (1.57 \times 3.74) - (0.79 \text{ X } 1.57)\ in^2 = 4.6315\ in^2$, see figure 3.2 and MD-439240

L_{sd}: Distance between the stiffening ring and downstream end flange, ~82.64 in
(See drawing ME-439317)

B: Factor determined from the applicable chart,
$= 0.75\ [PD_o \div (t + As/Ls)]$
$= 0.75\ [14.7 \text{ psi} \times 42.87 \text{ in} \div (0.31 \text{ in} + 4.6315\ in^2/82.64 \text{ in})]$
$= 1291$

Per figure CS-2 in Subpart 3 of section II, Part D, where B = 1084 falling below the left end of the material/temperature curve, so:

$A = 2B/E$
$= 2 \times 1291 / (28 \times 10^6)$
$= 0.0000922$
(per step 5 of UG-29, ASME VIII, Div. 1)

Then the Eq. 3-2 becomes:

$I_s' = \{[42.87^2 \times 82.64 \times (0.31 + 4.6315/82.64) \times 0.0000922] / 10.9\}in^4$
$= ((151{,}879 \times 0.366 \times 0.0000922) /10.9)\ in^4$
$= 0.4702\ in^4$

To find the combined moment of inertia of the cross section area of the stiffening ring and shell

It is found that from the figure 3.2:

$A_1 = 2.79 \text{ in} \times 0.31 \text{ in} = 0.8649\ in^2$
$A_2 \approx (1.57 \times 3.74) - (0.79 \text{ X } 1.57)\ in^2 = 4.6315\ in^2$
$y_1 = 0.155$ in
$y_2 = 2.18$ in
$y = (A_1 y_1 + A_2 y_2) \div (A_1 + A_2)$
$= 1.8613$ in
$dy_1^2 = (y - y_1)^2 = 2.9115\ in^2$
$dy_2^2 = (y - y_2)^2 = 0.1016\ in^2$
$I_1 = 0.0069\ in^4$
$I_2 = 6.8444\ in^4$
$I = \sum I_i + \sum A_i dy_i^2$
$= (6.8513 + 2.9887)\ in^4$
$= \underline{9.84\ in^4} > I' = 0.4702\ in^4$

Since the requirement moment of inertia I' is less than the value of I provided by the combined moment inertia of the end flange(downstream) and vessel shell, so the downstream end flange is satisfactory and it can be treated as a support of the vessel.

3.2 To Define the Upstream End Flange as the Stiffening Ring for the Vessel Shell Under the External Pressure

Reference drawings: Detail K of drawing ME-439317-2,
MD- 439240

L_{su}: Distance between the stiffening ring and upstream end flange, ~ 82.64 in = L_{sd}

A_{su}: ≈ (1.57 x 3.74) – (0.79 X 1.57) in^2 = 4.6315 in^2 = A_{sd}

Since using the same flange at the upstream end and the downstream end, also, $L_{su} = L_{sd}$; $A_{su} = A_{sd}$, $D_{ou} = D_{od}$, $t_u = t_d$, from the calculation steps of section 3.2, we can conclude that the requirement moment of inertia $I_u' = I_d'$, the available combined moment inertia $I_u = I_d$, that leads to the conclusion:

$$I_u = I_d = \underline{9.84\ \text{in}^4} > I_u' = I_d' = 0.4702\ \text{in}^4$$

So the upstream end flange is satisfactory and it can be treated as a support of the vessel.

4. Calculations and Analysis for the Vessel Shell Openings

4.1 Calculations for the Opening of Coldmass Port (#3) for the Reinforcement and Weld Size

The main vessel shell:

(ref. drawings: ME-439317, ME-4392620, MD-439252)

P: 14.7 psi, external design pressure

D_o: 42.87 in, outside dia. Of the vessel shell.

D_i: 42.24, inside dia. of the vessel shell.

t: 0.31 in, the nominal pipe wall thickness.

F_a: 15,000 psi, the max. allowable stress of the shell material at the operating temp, also know as S_v. at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Vessel shell material: SA 516, grade 60, carbon steel (see ME-439262)

L_v: Design length of the vessel section between lines of support (see Page UG-28(b)), ≈ 82.64"

Find out the required vessel shell thickness t_r:

1. Try t_n = 0.31"
   L_v/D_o = 82.64" / 42.87" = 1.93
   D_o/t_n = 42.87" / 0.31" = 138.29

   Per section UG-28, also Fig. G and Fig. CS-2 in Subpart 3 of Section II, part D, ASME VIII, Div. 1:
   It is found out that:
   A = 0.00043
   B = 6000
   Then:
   $P_a = (4B) / (3D_o/t_n)$
   $= (4 \times 6000) \div (3 \times 138.29)$
   $= 57.85$ psi
   where:
   P_a : The max. allowable external working pressure, psi
   A: Factor determined from Fig. G
   B: Factor determined from the applicable material chert in Fig. CS-2.

2. Try t_{r2} = 0.2
   L_v/D_o = 1.93
   D_o/t_{r2} = 42.87"/0.20" = 214

   Find:
   A ≈ 0.00023
   B = 3300
   Then:

$P_{a2} = (4B) / (3D_o/t_n)$
$= (4 \times 3300) \div (3 \times 214)$
≈ 19.64 psi

3. Try $t_{r3} = 0.175$
$L_v/D_o = 1.93$
$D_o/t_{r3} = 42.87"/0.175 = 245$

Find:
$A \approx 0.00019$
$B = 2800$

Then:
$P_{a3} = (4B) / (3D_o/t_n)$
$= (4 \times 2800) \div (3 \times 245)$
≈ 15.23 psi

So, $\mathbf{t_r = 0.175"}$

The coldmass port:

d_o: 25.197", outside dia. of the port.
D_i: 15.512", inside dia . of the port.
t_n: 4.842", wall thickness of the port.
t_{rn}: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
E: 28×10^6 psi, modulus of elasticity of the material at the operating temp.
Material: SA 182- 304L stainless steel pipe, $F_a = S_n = 16{,}700$ psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Reference drawings: ME-439317, MD-439251, MD-439252, and also as shown from figure 4.1.

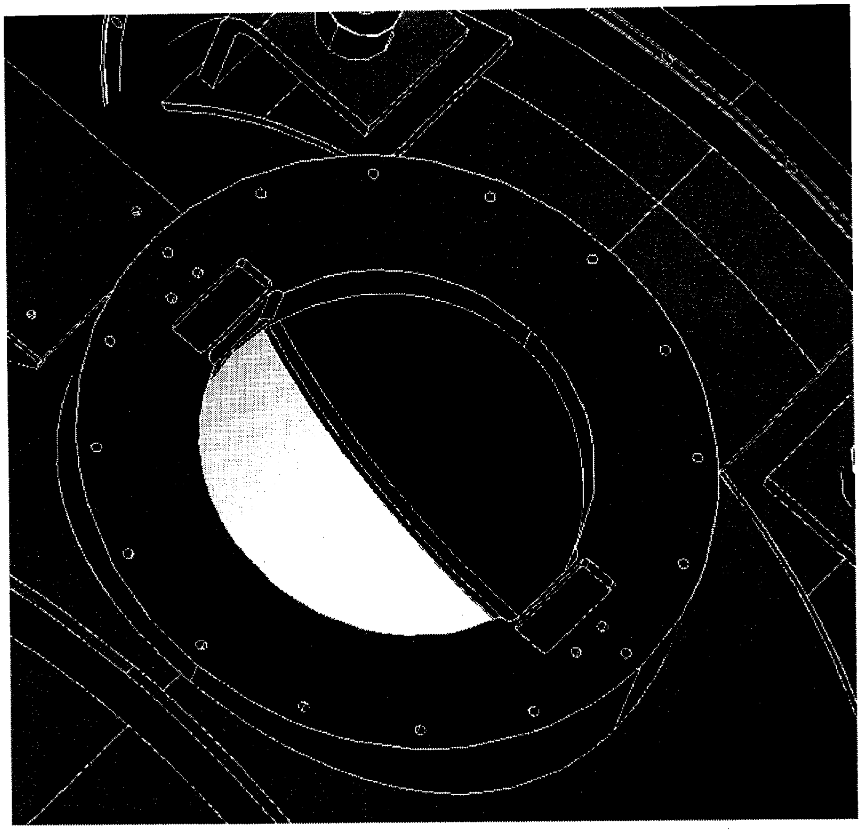

Figure 4.1, 3D view of the coldmass port #3 and the vessel shell

Find out the thickness required:

1. Assuming $L_1 = 5.49$ in, $t_{r1} = 0.375$ in, then:

   $L_1/d_o = 5.49/25.197 = 0.2179$
   $d_o/t_{r1} = 25.197/0.375 = 67$

   From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:

   $A = 0.0165$
   $B = 13800$
   $P_a = (4B)/(3(d_o/t_{r1})) = (4 \times 13800)/(3 \times 67) = 275$ psi $> P$

2. Try: $L_2 = 5.49$ in, $t_{r2} = 0.0625$ in

   $L_2/d_o = 0.2179$
   $d_o/t_{r2} = 403$

   get:
   $A = 0.00083$
   $B = 8250$

$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 8250)/(3 \times 403) = 27.30$ psi > P

3. Try: $L_3 = 6.64$ in, $t_{r3} = 0.0425$ in

   $L_3/d_o = 0.2179$
   $d_o/t_{r3} = 25.197/0.0425 = 593$

   get:
   A = 0.000475
   B = 6750
   $P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 6750)/(3 \times 593) = 15.18$ psi ≈ P
   So: $\mathbf{t_{rn} = t_{r3} = 0.0425}$ **in**

<u>*Size of the weld required*</u>:

Per UG-37, UW-16 and Fig. UW-16.1(p) of ASME VIII, Div.1.
t_o: the throat dimension of the outer attachment weld, $\geq 0.5\, t_{min}$.
t_i: the throat dimension of the inside of the shell cutout, $\geq 0.7 t_{min}$
$t_o = 0.5\, t_{min} = 0.5 \times 0.31" = 0.155"$, **so the fillet leg size is 0.22"**
$t_i = 0.7\, t_{min} = 0.7 \times 0.31" = 0.217"$, **so the fillet leg size is 0.31"**
(See details J & K of dwg. ME-439317-1 as the references)
So the sizes of the weld are satisfactory.

<u>*Area of the reinforcement required A_r*</u>:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1-f_{r1})$
$= 0.5 \times (16.299" \times 0.175" \times 1.0 + 2 \times 4.842" \times 0.25" \times 1.0 \times 0)$
$= \mathbf{1.4262\ in^2}$

Where:

d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.
   D = 16.299 in (per drawing ME-439262)
F: Correction factor
f_r: Strength reduction factor, not greater than 1.0
   f_{r1} =1.0 for nozzle wall abutting the vessel wall as shown in figure UG-40(j), (k), (n), (o) and (p).
   $f_{r2} = S_n / S_v = 16.7 / 15 = 1.113$

<u>*Area of reinforcement available A_a*</u>:

A_1: Larger of the following calculated value: (area available in shell):
$= d(E_1t - Ft_r)-2t_n(E_1t - Ft_r) \times (1-f_{r1})$
$= 16.299" \times (1 \times 0.31" - 1.0 \times 0.175") - 0$
$= 2.2004\ in^2$

or $= 2(t + t_n)(E_1 t - Ft_r) - 2t_n(E_1 t - Ft_r) \times (1-f_{r1})$
$= 2\ (0.31" + 4.842") \times (1.0 \times 0.31" - 1.0 \times 0.175") - 0$
$= 1.391\ in^2$
So pick $\mathbf{A_1 = 2.2004\ in^2}$

A_2: Smaller of the following:
$= 5\ (t_n - t_{rn}) t f_{r2}$
$= 5 \times (4.842\ in - 0.0425\ in) \times 0.31\ in \times 1.0$
$= 7.4392\ in^2$

or $= 5\ (t_n - t_{rn})\ f_{r2} t_n$
$= 5 \times (4.842\ in - 0.0425\ in) \times 1.113 \times 4.842\ in$
$= 129.33\ in^2$
So pick $\mathbf{A_2 = 7.4392\ in^2}$

A_3: Area available in inward nozzle, use the smallest value:
Since there is no inward nozzle
So pick $\mathbf{A_3 = 0.0\ in^2}$

A_{41}: Outward nozzle weld
$= leg^2\ f_{r2}$
$= 0.22^2\ in^2 \times 1.0$
$\mathbf{= 0.0484\ in^2}$
A_{43}: Inward nozzle weld
$= leg^2\ f_{r2}$
$= 0.31^2\ in^2 \times 1.0$
$\mathbf{= 0.0961\ in^2}$

Then:
$A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$
$= (2.2004 + 7.4392 + 0 + 0.0484 + 0.0961)\ in^2$
$= \mathbf{9.784\ in^2} > A_r = 1.4262\ in^2$

Per Fig. UG-37.1, section UG-37, the opening is adequately reinforced under the current designated boundary conditions.

The analysis and conclusion should be also true for coldmass port opening #2 (see drawing MD-439251), this is because: all basic data such as: d_o, D_i, L, t_o, t_i, S_n of coldmass #2 are the same as coldmass #3. So, per section UG-37 of ASME VIII, Div.1, the #3 opening is adequately reinforced under current design.

4.2 Calculations for the Opening of MC (main coupler) Port (#9) for the Reinforcement and Weld Size

The main vessel shell:

From section 4.1, it is found that $\mathbf{t_r = 0.175''}$

The MC port:

d_o: 11.339", outside dia. of the port.
D_i: 10.945", inside dia . of the port.
t_n: 0.197", wall thickness of the port.
t_m: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
E: 28×10^6 psi, modulus of elasticity of the material at the operating temp.
Material: SA 249-TP304L stainless steel pipe, $F_a = S_n$ = 14,200 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Reference drawings: ME-439317-2, and also as shown from figure 4.2.

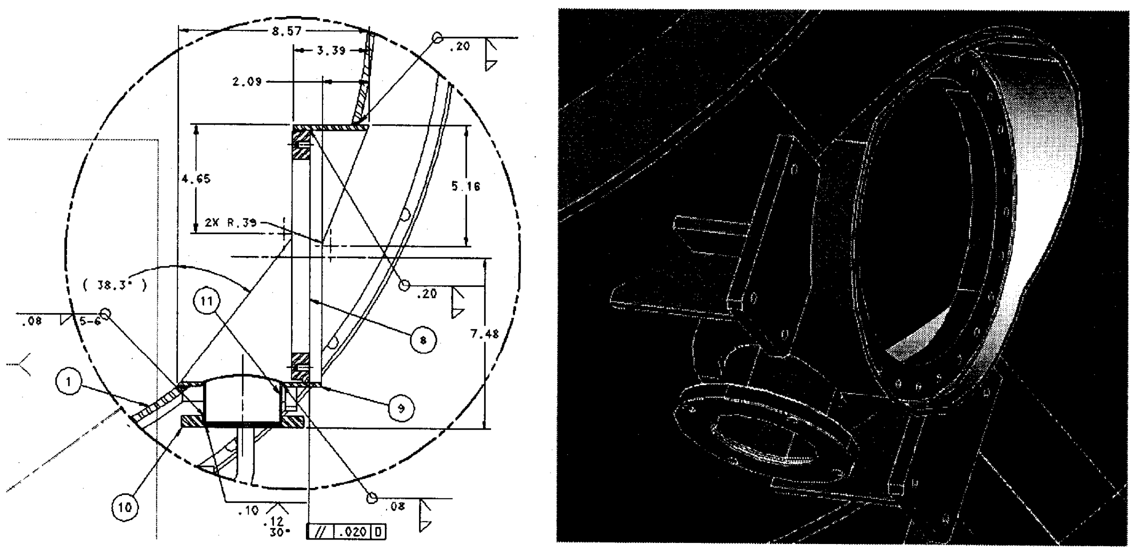

Figure 4.2, The partial 2D and 3D views of the MC port from ME-439317

Find out the wall thickness required:

1. Assuming L_1 = 8.57 in, t_{r1} = 0.125 in, then:

   L_1/d_o = 8.57/11.339 =0.7558

$d_o/t_{r1} = 11.339/0.125 = 91$

From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:

$A = 0.00225$
$B = 10500$
$P_a = (4B)/(3(d_o/t_{r1})) = (4 \times 10500)/(3 \times 91) = 153$ psi > P

2. Try: $L_2 = 8.54$ in, $t_{r2} = 0.0625$ in

$L_2/d_o = 0.7558$
$d_o/t_{r2} = 181$

get:
$A = 0.00075$
$B = 8050$
$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 8050)/(3 \times 181) = 59.3$ psi > P

Try: $L_3 = 8.54$ in, $t_{r3} = 0.033$ in

$L_2/d_o = 0.7558$
$d_o/t_{r3} = 344$

get:
$A = 0.000280$
$B = 3950$
$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 3950)/(3 \times 344) = 15.31$ psi $\approx$ P

So: $\mathbf{t_{rn} = t_{r3} = 0.033}$ **in**

Size of the weld required:

Per UG-37, UW-16 and Fig. UW-16.1© of ASME VIII, Div.1.
t_{min}: 0.197"
t_c: $\geq 0.7t_{min} = 0.138$", **so the fillet leg size is 0.195"**
Full penetration weld (bevel groove) plus the fillet weld t_c.

Area of the reinforcement required A_r:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1-f_{r1})$
$= 0.5 \times (11.339" \times 0.175" \times 1.0 + 2 \times 0.197" \times 0.175" \times 1.0 \times 0.053)$
$= \mathbf{0.994\ in^2}$

Where:
d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.

$D = 11.339$ in (per drawing ME-439317)

F: Correction factor

f_r : Strength reduction factor, not greater than 1.0

$f_{r1} = S_n / S_v$ for nozzle wall inserted through the vessel wall

$f_{r2} = S_n / S_v = 14.2 / 15 = 0.947$

Area of reinforcement available A_a:

A_1: Larger of the following calculated value: (area available in shell):

$= d(E_1t - Ft_r)-2t_n(E_1t - Ft_r) \times (1-f_{r1})$

$= 11.339" \times (1 \times 0.31" - 1.0 \times 0.175") - 2 \times 0.197 (1.0 \times 0.31 - 1.0 \times 0.175) \times 0.053$

$= (1.5308 - 0.0028)\ in^2$

$= 1.528\ in^2$

or $= 2(t + t_n)(E_1t - Ft_r)-2t_n(E_1t - Ft_r) \times (1-f_{r1})$

$= 2 (0.31" + 0.197") \times (1.0 \times 0.31" - 1.0 \times 0.175") - 2 \times 0.197 (1.0 \times 0.31 - 1.0 \times 0.175) \times 0.053$

$= (0.1825 - 0.0028)\ in^2$

$= 0.1341\ in^2$

So pick **$A_1 = 1.528\ in^2$**

A_2: Smaller of the following:

$= 5 (t_n-t_{rn})tf_{r2}$

$= 5 \times (0.197\ in - 0.033\ in) \times 0.31\ in \times 0.947$

$= 0.2407\ in^2$

or $= 5 (t_n-t_{rn}) f_{r2}t_n$

$= 5 \times (0.197\ in - 0.033\ in) \times 0.947 \times 0.197\ in$

$= 0.153\ in^2$

So pick **$A_2 = 0.153\ in^2$**

A_3: Area available in inward nozzle, use the smallest value:

Since there is no inward nozzle

So pick **$A_3 = 0.0\ in^2$**

A_{41}: Outward nozzle weld

$= leg^2 f_{r2}$

$= 0.195^2\ in^2 \times 0.947$

$= 0.036\ in^2$

A_{43}: Inward nozzle weld

$= leg^2 f_{r2}$

$= 0.0^2\ in^2 \times 1.0$

$= 0.0\ in^2$

Then:

$A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$

$= (1.528 + 0.153 + 0 + 0.036 + 0.0)\ in^2$

$= \mathbf{1.717\ in^2} > A_r = 0.994\ in^2$

Per Fig. UG-37.1, section UG-37, the opening is adequately reinforced under the current designated boundary conditions.

The analysis and conclusion should be true for all 2 openings (#9 openings, see drawings ME-439317 & ME-439262)

4.3 Calculations for the Opening of Electronic Port (#11 opening) for the Reinforcement and Weld Size

The main shell (Tube #9)

From section 4.2, it is found that: **t_r = 0.033 in**

The instrumentation port:

d_o: 3.50", outside dia. of the port.
D_i: 3.334", inside dia . of the port.
t_n: 0.083", wall thickness of the port.
t_m: required thickness of the port.
L: length of projection defining the thickness portion of integral reinforcement of the port beyond the outside surface of the vessel wall.
E: 28 x 10^6 psi, modulus of elasticity of the material at the opening temp.
Material: SA 249-TP304L stainless steel pipe, $F_a = S_n$ = 14,200 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
Reference drawings: section D-D, ME-439317, and also as shown from figure 4.2.

Find out the wall thickness requires:

Assuming L_l = 2.0075 in, t_{rl} = 0.083 in, then:
where: L_l = 7.48 – (11.339/2 – 0.197) = 2.0075"

L_l/d_o = 2.0075/3.50 =0.5736
d_o/t_{rl} = 3.50/0.083 = 42

From Fig. G and Fig. HA-3 of subpart 3 of Sect. II, part D:

A = 0.009
B = 13200
$P_a = (4B)/(3(d_o/t_{r1}))$ = (4 x 13200)/(3 x 42) = 419 psi > P

Try: L_2 = 2.0075 in, t_{r2} = 0.015625 in

L_2/d_o = 0.5736
d_o/t_{r2} = 224

get:
A = 0.0007
B = 8000
$P_a = (4B)/(3(d_o/t_{r2}))$ = (4 x 8000)/(3 x 224) = 47.62 psi > P

Try: L_3 = 2.0075 in, t_{r3} = 0.0094 in

$L_3/d_o = 0.5732$
$d_o/t_{r3} = 372$

get:
$A = 0.0003$
$B = 4250$
$P_a = (4B)/(3(d_o/t_{r2})) = (4 \times 4250)/(3 \times 372) = 15.23$ psi $\approx P$

So: $\mathbf{t_{rn} = t_{r3} = 0.0094}$ **in**

Size of the weld required:

Per UG-37, UW-16 and Fig. UW-16.1(c) of ASME VIII, Div.1.
t_{min}: 0.083"
$t_{c:}$ $\geq 0.7t_{min} = 0.056$", **so the fillet leg size is 0.08"**
Full penetration weld (bevel groove) plus the fillet weld t_c.

Area of the reinforcement required A_r:

Per section UG-37(d)(1):

$A_r = 0.5(dt_rF + 2t_nt_rF(1\text{-}f_{r1})$
$= 0.5 \times (3.5" \times 0.033" \times 1.0 + 2 \times 0.083" \times 0.033" \times 1.0 \times 0)$
$= \mathbf{0.0578\ in^2}$

Where:
d: Finished diameter of circle opening, see Fig. UG-37.1 of UG-37.
D = 11.339 in (per drawing ME-439262)
F: Correction factor
f_r: Strength reduction factor, not greater than 1.0
$f_{r1} = S_n / S_v$ for nozzle wall inserted through the vessel wall
$f_{r2} = S_n / S_v = 14.2 / 14.2 = 1.0$

Area of reinforcement available A_a:

A_1: Larger of the following calculated value: (area available in shell):
$= d(E_1t - Ft_r)\text{-}2t_n(E_1t - Ft_r) \times (1\text{-}f_{r1})$
$= 3.50" \times (1 \times 0.197" - 1.0 \times 0.033") - 2 \times 0.083 (1.0 \times 0.197 - 1.0 \times 0.033) \times 0$
$= 0.574\ in^2$

or $= 2(t + t_n)(E_1t - Ft_r)\text{-}2t_n(E_1t - Ft_r) \times (1\text{-}f_{r1})$
$= 2 (0.197" + 0.083") \times (1.0 \times 0.197" - 1.0 \times 0.033") - 2 \times 0.197 (1.0 \times 0.375 - 1.0 \times 0.25) \times 0.0$
$= (2 \times 0.28 \times 0.164)\ in^2$
$= 0.0918\ in^2$

So pick $\mathbf{A_1 = 0.574\ in^2}$

Since $A_1 = 0.574$ in^2
Also: $A_a = A_1 + A_2 + A_3 + A_{41} + A_{43}$

That leads to:
$A_a > A_r = 0.0578$ in^2

Per Fig. UG-37.1, section UG-37, the opening is adequately reinforced under the current designated boundary conditions.

The analysis and conclusion should be true for all 2 electronic ports (#11 openings, see section D-D, drawing ME-439317 and Figure 4.2 of Page 19).

5. Analysis and Calculations for the Flanges,Their Bolts and Welds

5.1 Downstream End Flange and the Bellow Flange

References: ME-439317-2, MD-439240
Endcap (module 2-3), #0988348/0, DESY – MKS1
and Figure 5.1

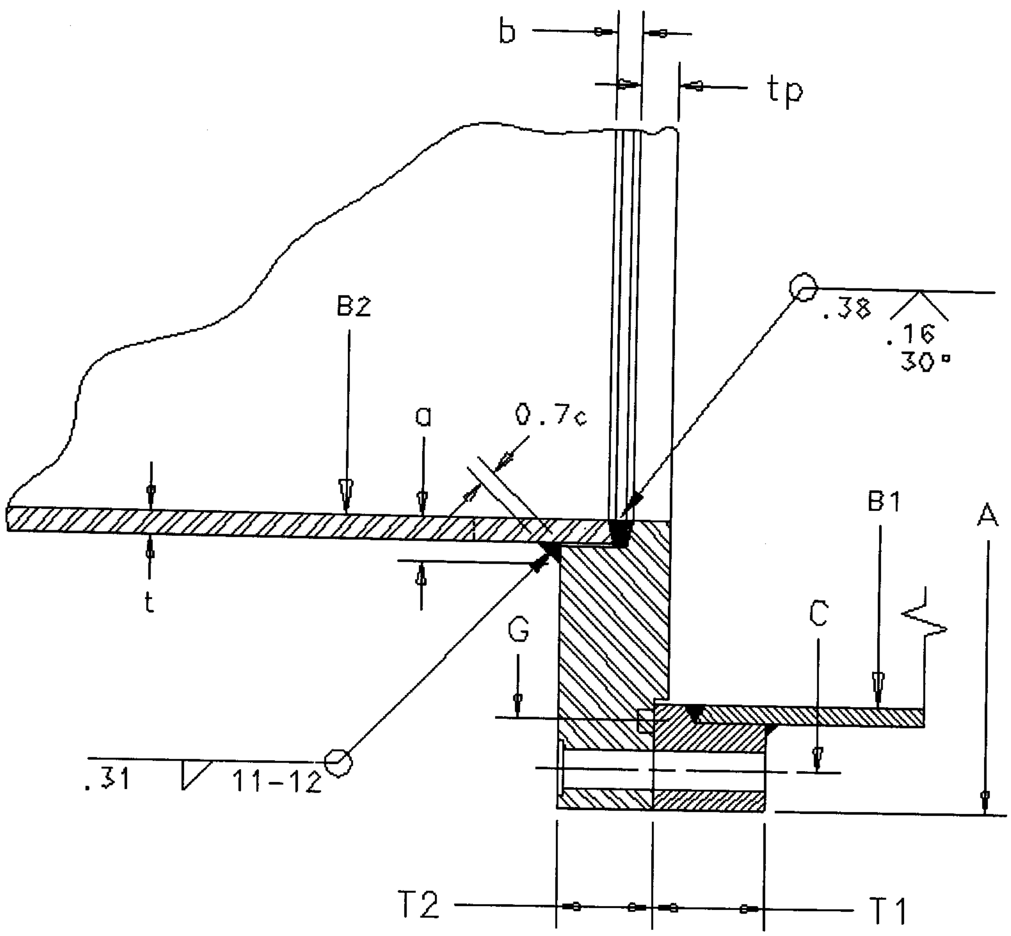

Figure 5.1, Downstream end flanges with the below flange

Figure 5.1 is the assembly configuration of the downstream flange connecting with the below flange per DESY's layout "Endcap (module 2-3)"

It was found that:

A = 50.35" (dia.)
B_1 = 47.34" (dia.)
B_2 = 42.24" (dia.)

$C = 49.21"$ (dia.)
$G = 47.889"$ (dia.)
$T_1 = 1.575"$
$T_2 = 1.575"$
$P = 14.7$ psi
$n = 24$
$E_1 = E_2 = 2.8 \times 10^7$ psi

Material: SA 182-F304L stainless steel, $F_a = S_f = 16{,}700$ psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).
assuming bolt spec: ½ - 13 UNC
$A_b = 24 \times\ 0.1257\ in^2 = 3.0168\ in^2$
The cross-sectional area of the bolts using the root dia. (0.4001") of the thread in sq. in.

Per section Y-5.1, Y-5.2, figure Y-5.1.2 of Appendix Y,
Section 2-4, figure 2-4 of Appendix 2 of ASME VIII:

$B_2 > ½ C$,
So it is defined as: Class 2, Category 3 flange

Flange Analysis:

The moment due to the pressure M'_p:

$M'_p = H_D'h_D' + H_T'h_T'$ (per Y-6.2(3), Appendix Y, ASME VIII)

$H_D' = 0.785\ B_2^2\ P$
$= 0.785 \times 42.24^2\ in^2 \times 14.7$ psi
$= 20{,}589$ lbs.

$H_T' = 0.785P\ (B_1^2 - B_2^2)$
$= 0.785 \times 14.7$ psi $(47.34^2 - 42.24^2)\ in^2$
$= 5{,}272$ lbs.

$h_D' = (B_1 - B_2) \div 2$
$= (47.34 - 42.24)$ in $\div 2$
$= 2.55$ in

$h_T' = (B_1 - B_2) \div 4$
$= (47.34 - 42.24)$ in $\div 4$
$= 1.275$ in

So: $M'_p = (20{,}589 \times 2.55)$ lbs-in + $(5{,}272 \times 1.275)$ lbs-in
$= 59{,}224$ lbs-in

$= C_5$

(per section Y-6.2(4)(a), Appendix Y, ASME VIII, Div.1)

Let: $C_6 = 0.829 / \log(B_1/B_2)$
$= 0.829 \div 0.0495$
$= 16.747$

(per section Y-6.2(4)(b), Appendix Y, ASME VIII, Div.1)

$C_l = [1 - 2.095 J_s \log(A/B_1)] \div (- C_6 - 1.738\ J_s)$

(per section Y-6.2(4)(c), Appendix Y, ASME VIII, Div.1)

Where:

$J_s = (1/B_1)\ [2h_D/\beta + h_C/\alpha] + \pi\ r_B$

$h_D = (C - B) / 2$
$= (49.21 - 47.34) / 2$ in
$= 0.935$ in

$h_C = (A - C) / 2$
$= (50.35 - 49.21) / 2$ in
$= 0.57$ in

$\beta = (C + B)/ 2B$
$= (49.21 + 47.34) / (2 \times 47.34)$
$= 1.02$

$\alpha = (A + C)/ 2B$
$= (50.35 + 49.21) / (2 \times 47.34)$
$= 1.052$

$r_B = (1/n)[4/(1 - (\overline{AR})^2)^{1/2}\ \tan^{-1}((1+ \overline{AR})/(1 - \overline{AR}))^{1/2} - \pi - 2\overline{AR}]$

where:

$\overline{AR} = nD/ \pi C$
$= (24 \times 0.551") / \pi\ 49.21"$
$= 0.0855$

D: the bolt hole diameter

$r_B = (1/24)[4/(1 - (0.0855)^2)^{1/2}\ \tan^{-1}((1+ 0.0855)/(1 - 0.0855))^{1/2} - \pi - 2 \times 0.0855]$
$= 0.041667 \times (4.0147 \times 0.8282 - \pi - 0.171)$
$= 0.0005159$

$J_s = (1/47.34")\ [2 \times 0.935/1.02 + 0.57/1.052] + \pi \times 0.0005159$
$= 0.0211 \times (1.8333 + 0.5418) + 0.0016$
$= 0.0517$

So:

$C_l = [1 - 2.095 \times 0.0517 \times \log(50.35/47.34)] \div (- 16.747 - 1.738 \times 0.0517)$
$= (1 - 0.0029) \div (-16.8369)$
$= -0.0592$

$C_2 = (1.738 J_p M_p - C_5 C_6) \div (- C_6 - 1.738\, J_s)$
(per section Y-6.2(4)(d), Appendix Y, ASME VIII, Div.1)

where:

$J_p = (1/B_1)\,[h_D/\beta + h_C/\alpha] + \pi\, r_B$
$= (1/47.34") [0.935/1.02 + 0.57/1.052] + \pi \times 0.0005159$
$= 0.0211 \times (0.9167 + 0.5418) + 0.0016$
$= 0.0324$

$M_p = H_D h_D + H_T h_T + H_G h_G$ (per sectionY-3, Appendix Y, ASME VIII)

$H_D = 0.785\, B_1^2\, P$
$= 0.785 \times 47.34^2\ in^2 \times 14.7$ psi
$= 25{,}861$ lbs.

$H\ = 0.785 G^2 P$
$= 0.785 \times 47.889^2\ in^2 \times 14.7$ psi
$= 26{,}464$ lbs.

$H_T = H - H_D$
$= 26{,}464 - 25{,}861$
$= 603$ lbs.

$h_G\ = (C - G)/2$
$= (49.21 - 47.889)$ in / 2
$= 0.6605$ in

$h_D = 0.935$ in

$h_T = (h_D\ + h_G) / 2$
$= 0.7978$ in

$H_G = W - H$, gasket load
(per section 2-3, appendix 2, ASME VIII)
$W = H + H_p$,
flange design load for the operating condition
(per section 2-5(c)(1), appendix 2, ASME VIII)

which leads to:

$H_G = W - H$
$=\ H + H_p - H$
$=\ H_p$
$= 2b \times 3.14 GmP$
$= 2 \times 0.281$ in $\times 3.14 \times 47.889$ in $\times 3.5 \times 14.7$ psi
$=\ 4{,}348$ lbs.

$M_p = H_D h_D + H_T h_T + H_G h_G$
$= (25,861 \times 0.935 + 603 \times 0.7978 + 4,348 \times 0.6605)$ lbs-in
$= (24,180 + 481 + 2,872)$ lbs-in
$= 27,533$ lbs-in

$C_2 = (1.738 J_p M_p - C_5 C_6) \div (- C_6 - 1.738\ J_s)$
$= (1.738 \times 0.0324 \times 27,533 - 16.747 \times 59,244)$ lbs-in $\div (-16.747 - 1.738 \times 0.0517)$
$= 990,609$ lbs-in / (-16.8369)
$= -58,836$ lbs-in

For the rigid body rotation of the flange times E*:

$E_1{}^* \theta_{rbi} = X(C_4 - C_2) \div (1.206 \log(A/B_1) - XC_3 - (1 - X)C_1)$

where:

$C_3 = -(0.575 - 1.206\ Js \log(A/B_1)) \div (Js + t_1^3/F_1')$
$= 0$

$C4 = -(J_p M_p) \div (Js + t_1^3/F_1')$
$= 0$
$\because F_1' = 0$ (per section Y-3(6c), Appendix Y, ASME VIII, Div.1)

$X = E_1{}^* / (E_1{}^* + E_2{}^*)$
$E_1{}^* = E_1\ t_1^3$
$E_2{}^* = E_2\ t_2^3$
$\because E_1 = E_2$
$\therefore X = E_1\ t_1^3 / (E_1\ t_1^3 + E_2\ t_2^3)$
$= t_1^3 / (t_1^3 + t_2^3)$
$= 3.907 / (3.907 + 3.907)$
$= 0.50$

$E_1{}^* \theta_{rbi} = 0.50(0 + 58,836) \div [(1.206 \times 0.0268) - (0.50 \times 0.0) - (1 - 0.50) \times -0.0592)]$
$= 0.50 \times (+58,836)$ lbs-in $\div (0.0323 - 0.0 + 0.0296)$
$= 29,418$ lbs-in $\div 0.0619$
$= 475,250$ lbs-in

$E_2{}^* \theta_{rbii} = -E_1{}^* \theta_{rbi} (E_2{}^* / (E_1{}^*)$
$= -(475,250 \times 1.0)$ lbs-in
$= -475,250$ lbs-in

Total Flange Moment at Diameter B_1:

$M_{s1} = C_3 (E_1{}^* \theta_{rbi}) + C_4$
$= 0.0$

$M_{s2} = C_1 (E_2 * \theta_{rbii}) + C_2$
$= - 0.0592 \times - 475{,}250$ lbs-in - 58,836 lbs-in
$= - 30{,}701$ lbs-in

Unbalanced Flange Moment at Diameter B_1:

$M_{u1} = 1.206 (E_1 * \theta_{rbi}) \log(A/B_1)$
$= 1.206 \times 475{,}250$ lbs-in $\times 0.0268$
$= 15{,}360$ lbs-in
$M_{u2} = 1.206 (E_2 * \theta_{rbii}) \log(A/B_1)$
$= 1.206 \times (- 475{,}250$ lbs-in$) \times 0.0268$
$= -15{,}360$ lbs-in

Balanced Flange Moment at Diameter B_1:

$M_{b1} = M_{s1} - M_{u1}$
$= 0.0$ lbs-in - (15,360 lbs-in)
$= -15{,}360$ lbs-in
$M_{b2} = M_{s2} - M_{u2}$
$= -30{,}701$ lbs-in + 15,360 lbs-in
$= -15{,}341$ lbs-in

Slope of Flange at Diameter B_1 times E:

$E_1 \theta_{Bi} = (5.46/\pi t_1^3) (J_s M_{b1} + J_p M_p) + E_1 * \theta_{rbi} / t_1^3$
$= 0.4448$ in^{-3} x (0.0517 x (– 15,360 lbs-in) + 0.0324 x 27,533 lbs-in) +
475,250 lbs-in / 3.907 in^3)
= (44 + 121,641) psi
= 121,685 psi
$E_2 \theta_{Bii} = (5.46/\pi t_2^3) (J_s M_{b2} + J_p M_p) + E_2 * \theta_{rbii} / t_2^3$
(per section Y-6.2(e), Appendix Y, ASME VIII, Div.1)
= [0.4448 x (0.0517 x (-15,341) + 0.0324 x 27,533) - 475,250 / 3.907] psi
= (44 – 121,641) psi
= -121,597 psi

Contact Force between Flange at h_c:

$H_c = (M_p + M_{b1}) / h_c$
= (27,533 - 15,360) lbs-in / 0.57 in
= 12,173 lbs.

Bolt Load at Operating Conditions:

$W_{m1} = H + H_G + H_C$
= (26,464 + 4,348 + 12,173) lbs

$= 42{,}985$ lbs.

Operating Bolt Stress f_b:

$f_b = W_{m1} / A_{b1}$
$= 42{,}985$ lbs / $3.0168\ in^2$ (See page 52, section 5.1 of the note)
$= \underline{14{,}249\ psi}$

Design Pre-stress in Bolts:

$S_i = f_b - 1.159\ h_c^2\ (M_p + M_{b1}) / (2(1 - X)a\ t_1^3\ l\ r_{e1}B_1)$
where:
r_{e1}: ≈ 1.0, elasticity factor, the ratio between E_{flange} and E_{bolt}
$a = (A + C) / 2B_1$
$= (50.35 + 49.21)$ in / 2 x 47.34 in
$= 1.052$ (shape factor)
l: $= t_1 + t_2 + 0.5d_b$ (calculated strain length of the bolt)
$= (1.575 + 1.575 + 0.25)$ in
$= 3.40$ in

$S_i = 14{,}249$ psi– $1.159 \times 0.57^2\ in^2 \times (27{,}533 - 15{,}360)$lbs-in / $(2(1 - 0.50) \times 1.052 \times 1.575^3\ in^3 \times 3.40$ in x 1.0 x 47.34 in)
$= 14{,}249$ psi – 4,584 lbs-in^3 / 662 in^5
$= 14{,}242$ psi

Redial Stress in Flange I @ Bolt Circle:

$S_{R1} = 6(M_p + M_{S1}) / t_1^2(\pi C - nD)$
$= 6 \times (27{,}533 - 0)$ lbs-in / $1.575^2\ in^2 \times (\pi\ 49.21$ in – 24 x 0.551 in)
$= 165{,}198$ lbs-in / 350.7 in^3
$= 471$ psi $< S_n = 16{,}700$ psi

Tangential Stress in Flange I @ Inside Diameter:

$S_{t1} = t_1\ E_1\ \theta_{Bi} / B_1 + (2Ft_1\ Z/(ho + Ft_1) - 1.8)(M_{s1}/(\pi B_1 t_1^2)$
$= 1.575$ in x 121,685 psi / 47.34 in
$= 4{,}048$ psi $< S_n = 16{,}700$ psi

Tangential Flange Stress Adjacent to Central Nozzle:

$S_{t2n} = Y(M'_p + M_{S2}) / t_2^2\ B_2$
where:
$Y = (1/(K-1))[0.66845 + 5.71690\ K^2\log_{10}K/(K^2 - 1)]$
(per figure 2-7.1, Appendix 2, ASME VIII, Div. 1)
$K = A/B$
$= 50.35 / 42.24$

$= 1.192$

$Y = (1/(1.192-1))[0.66845 + 5.71690 \times 1.4209 \times \log_{10}K/(K^2 -1)]$
$= 5.2083 \times (0.6685 + 5.7169 \times 0.2575)$
$= 11.1489$

$\therefore\ S_{t2n} = 11.1489 \times (59{,}224 - 30{,}701)$ lbs-in / $(1.575^2 \times 42.24)$ in^3
$= 3{,}035$ psi $< S_n = 16{,}700$ psi

Longitudinal Hub Stress in Flange 1:

$S_{h1} = 0$
(per equation (34c), section Y-6.3, appendix Y, ASME VIII, Div.1)

Radial Stress in Flange 2 @ bolt circle:

$S_{r2b} = 0$

Radial Stress in Flange 2 @ Diameter B_1:

$S_{r2B1} = 0$

Tangential Stress at Center of Flange 2:

$S_{t2B2} = (0.3094\ PB_1^2 / t_2^2) - (6\ M_{S2} / \pi B_1\ t_2^2)$
$= (0.3094 \times 14.7$ psi $\times 47.34^2$ in^2 / 1.575^2 in^2)
$-\quad (6 \times (-30{,}701$ lbs-in) / π 47.34 in $\times 1.575^2$ in^2)
$= 4{,}109$ psi $+ 499$ psi

$= 4{,}608$ psi $< S_n = 16{,}700$ psi

<u>Welds Analysis and Calculations:</u>
See UW-13 (e)(2), Fig. UW-13.2, section 2-3 and Fig. 2-4(4a), Appendix 2 of ASME VIII, Div.1. for the related codes and the physical representation of the nomenclatures.

Per above codes, it is required that the throat of the fillet $c = 0.7t_{min.} = 0.7 \times 0.31$"
$= 0.22$",
So the fillet leg size is ~ 0.31" as shown in figure 5.1.

It is found that:
$t_p = 0.53" > 0.25"$
where is the smallest of t_n, t_x and 0.25" is 0.25"

$c = t_n = 0.31"$ = leg of the fillet
$b = 0.16" + 2 \times \tan 15^o \times 0.38"$

$= 0.36"$

$a = 0.38" + 0.31" - 0.06"$

$= 0.63"$

$\therefore$ $a + b = 0.99" \geq 3t_n = 0.93"$

The weld size design for the end flange is satisfactory.

__Also run a weld strength calculation for the noted location as the 2nd analysis approach__:

Radial pressure load F_{rp}:

$F_{rp} = P\ L_{sd}$

$= 14.7$ psi x 82.84 in

$= 1,215$ lbs/in

Shear flow due to radial shear load F_{sr}:

$F_{sr} = VQ/I$

where:

Radial shear load V:

$V = 0.01\ P\ L_{sd}\ D_o$

$= 0.01$ x 14.7 psi x 82.64 in x 42.87 in

$= 521$ lbs.

The 1st moment of area Q:

$Q = 2.79$ in x 0.31 in x $(1.8613 - 0.155)$ in

$= 1.4758\ in^3$

$I = 9.84\ in^4$

$F_{sr} = VQ/I$

$= 521$ lbs. x $1.4758\ in^3 / 9.84\ in^4$

$= 78$ lbs/in

The required resultant weld load F_{rw}:

$F_{rw} = (1,215^2 + 78^2)^{1/2}$ lbs/in

$= 1,218$ lbs/in

The welds for connecting the flange and the vessel shell is single groove weld inside and intermittent fillet outside, the lowest allowable weld stress F_{aw}:

$F_{aw} = 0.60\ S_n = 0.60$ x 16.7 ksi

$= 10.02$ ksi

The requires calculated minimum weld size C_m:

$C_m = F_{rw} / F_{aw}$

$= 1,218 / 10,020$ (in)

$= 0.122$ in < 0.38 in (designated single groove weld)

If adding the value of the outside intermittent fillet weld (0.31"), the weld

size even can go further smaller size.
So, the design of the weld for the upstream end flange is satisfactory.

Per the above calculations and analysis, the downstream end flange and the bellow flange have met the codes of ASME VIII, Div.1.

5.2. Upstream End Flange, Slide Flange and Bellow Flange

References: ME-439317-2, MD-439240
Endcap (module 2-3), #0988348/0, DESY – MKS1
and Figure 5.2

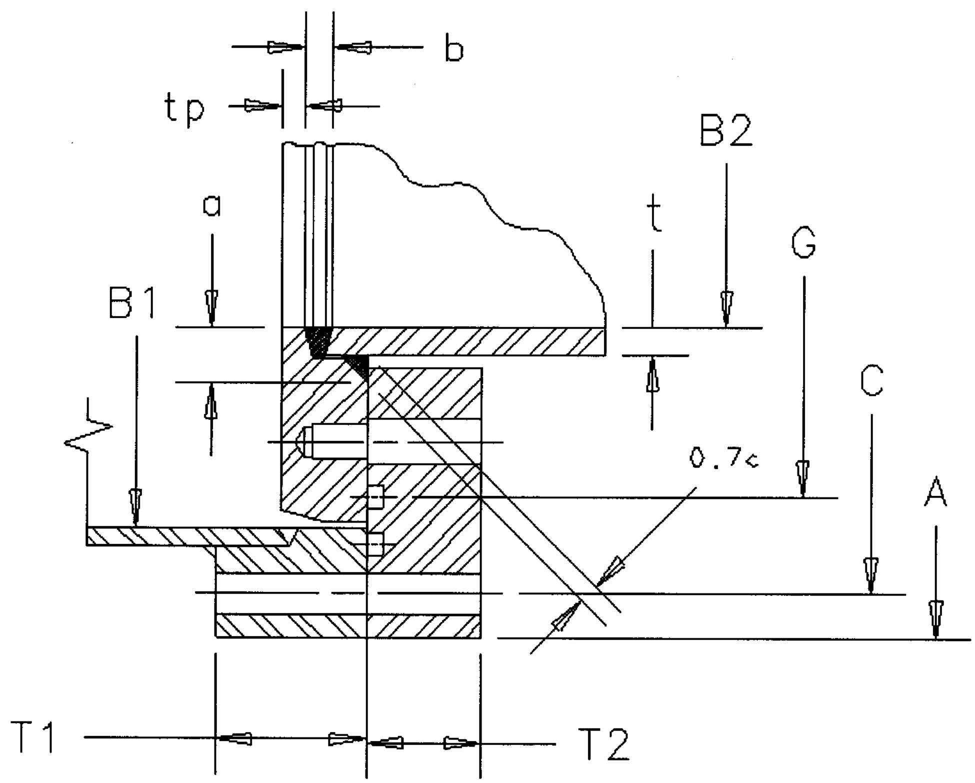

Figure 5.2. Upstream flange with the below flange

Since there some information are missing at this moment, so it is to make the best assumption as followings:

1. The upstream end flange connect to the downstream stream end flange of the bellow (see drawing Endcap (module 2-3), #0988348/0, DESY – MKS1).
2. The above two flanges are connected by a slide flange.
3. A = 50.35", G = 47.889", T_1 = 1.57", T_2 = 1.57", B_1 = 47.34", B_2 = 42.24", C = 49.21"
4. n = 24, D = 0.551"
   E = 28 x 10^6 psi, Modules of elasticity of the flange materials

Material: SA 182-F304L stainless steel, F_a = S_f = 16,700 psi at the operating temp., -20^0 F to 100^0 F (per Table 1A, part D, Section II, ASME Boiler &

Pressure Vessel Code 1995 edition).
S_b = 25 ksi for bolt material: SA-193, S304 for operating temperature, -20^0 F to 100^0 F (per Table 3, part D, Section II, ASME Boiler & Pressure Vessel Code 1995 edition).

Per section Y-5.1, Y-5.2, figure Y-5.1.2 of Appendix Y,
Section 2-4, figure 2-4 of Appendix 2 of ASME VIII:

$B_2 > ½ C$,
So it is defined as: Class 2, Category 3 flange

In order to reduce the amount of work required as shown on section 5.1, per section Y-9, of Appendix Y, ASME VIII, Div.1, we apply the method of estimating flange thickness and bolting for the downstream flanges.

$t_a = 2.45\ [Mp / ((\pi C - nD)S_f)]^{1/2}$
(per eq. 39 of Y-9, appendix Y, ASME VIII, Div.1)

where:
$M_p = H_D h_D + H_T h_T + H_G h_G$ (per sectionY-3, Appendix Y, ASME VIII)

$H_D = 0.785\ B_1^2\ P$
$= 0.785 \times 47.34^2\ in^2 \times 14.7$ psi
$= 25{,}861$ lbs.

$H\ = 0.785 G^2 P$
$= 0.785 \times 47.889^2\ in^2 \times 14.7$ psi
$= 26{,}484$ lbs.

$H_T = H - H_D$
$= (26{,}464 - 25{,}861)$ lbs.
$=\ 603$ lbs.
$h_G = (C - G)/2$
$= (49.21 - 47.889)$in / 2
$= 0.6605$ in

$h_D = (C - B) / 2$
$= (\ 49.21 - 47.34) / 2$
$= 0.935$ in

$h_T = (h_D\ + h_G) / 2$
$= 0.7978$ in

$H_G = W - H$, gasket load
(per section 2-3, appendix 2, ASME VIII)
$W = H + H_p$,

flange design load for the operating condition
(per section 2-5(c)(1), appendix 2, ASME VIII)

which leads to:

$$
\begin{aligned}
H_G &= W - H \\
&= H + H_p - H \\
&= H_p \\
&= 2b \times 3.14GmP \\
&= 2 \times 0.281 \text{ in} \times 3.14 \times 47.889 \text{ in} \times 3.50 \times 14.7 \text{ psi} \\
&= 4{,}348 \text{ lbs.}
\end{aligned}
$$

$$
\begin{aligned}
M_p &= H_D h_D + H_T h_T + H_G h_G \\
&= (25{,}861 \times 0.935 + 603 \times 0.7978 + 4{,}348 \times 0.6605) \text{ lbs-in} \\
&= (24{,}180 + 481 + 2{,}872) \text{ lbs-in} \\
&= 27{,}533 \text{ lbs-in}
\end{aligned}
$$

$$
\begin{aligned}
t_a &= 2.45 \, [Mp / ((\pi C - nD)S_f)]^{1/2} \\
&= 2.45 \times [\, 27{,}533 \text{ lbs-in} / ((\pi \; 49.21 \text{ in} - 24 \times 0.551 \text{ in}) \times 16{,}700 \text{ psi})]^{1/2} \\
&= 2.45 \times 0.1078 \text{ in} \\
&= 0.264 \text{ in}
\end{aligned}
$$

$$
\begin{aligned}
t_b &= 0.56 \, B_1 \, (P/S_f)^{1/2} \\
&= 0.56 \times 47.34 \text{ in} \times (14.7 \text{ psi} / 16{,}700 \text{ psi})^{1/2} \\
&= 0.56 \times 47.34 \text{ in} \times 0.0297 \\
&= 0.788 \text{ in}
\end{aligned}
$$

Since t_c is the greater of t_a or t_b, so:

$t_c = t_b = 0.788"$

Per equation 42 of A-7, Appendix Y, ASME VIII, Div. 1:

$$
\begin{aligned}
A'_b &= (H + 2\,M_p / (A - C)) \div S_b \\
&= (26{,}464 \text{ lbs} + 2 \times 27{,}533 \text{ lbs-in} / (50.35 - 49.21) \text{ in}\,) \div 25{,}000 \text{ psi} \\
&= (26{,}464 \text{ lbs} + 48{,}304 \text{ lbs}) / 25{,}000 \text{ psi} \\
&= 2.9907 \text{ in}^2
\end{aligned}
$$

Per Table Y-9.1, Appendix Y, ASME VIII, Div.1, it is found:

$t_1 = 1.1t_a = 0.29" < T_1 = 1.57"$

$t_2 = 1.1t_c = 0.867" < T_2 = 1.57"$

So the thickness of the flanges are ok for the upstream end flanges.

Welds Analysis and Calculations:

The weld configurations are the same as the upstream flange, it designed per section UW-13 (e)(2), Fig. UW-13.2, section 2-3 and Fig. 2-4(4a), Appendix 2 of ASME VIII, Div.1

Also, since the valus of L_{sd} for upstream end flange is the same as for the downstream end flange), "I" value for upstream end flange is similar as the downstream flange, all these lead to a similar values of F_{rw} and C_m for the upstream end and downstream end flanges, so the same sizes and types of the weld design of the upstream end flange are satisfactory.

Per the above calculations, the upstream end flange and the bellow flange have met the codes of ASME VIII, Div.1.

5.3. Flanges #8, #10, #2 and #3

In order to reduce the amount of work required as shown on section 5.1, per section Y-9, of Appendix Y, ASME VIII, Div.1, we apply the method of estimating flange thickness and bolting for the above flanges.

Since the information of the most mating flanges are not available at this moment, I made some engineering judgments per the best of my logic knowledge.

Table 5.3, Estimated Calculated Flange thickness and bolting VS. the designated data

		Flanges (see drawing ME-439317)			
nomenclature	unit	#8	#10	#2	#3
		MC-439243	MC-439246	MC-439251	MC-439252
P	psi	14.7	14.7	14.7	14.7
Sf	psi	18,800	18,800	18,800	18,800
A	in	10.945	5.512	25.2	25.2
B1	in	8.386	3.334	21.65	21.65
B2	in	3.75	3.334	0	0
C	in	9.646	4.882	23.622	23.622
G	in	8.793	4.012	22.636	22.636
Classification		3	1	3	3
Categorization		3	3	3	3
n		24	4	16	16
D	in	0.3125	0.394	0.394	0.394
m		3.5	3.5	3.5	3.5
b		0.21	0.21	0.21	0.21
HD	lbs	812	128	5409	5409
H	lbs	892	186	5913	5913
HT	lbs	81	57	504	504
hG	in	0.4265	0.435	0.493	0.493
hD	in	0.63	0.774	0.986	0.986
hT	in	0.5283	0.6045	0.7395	0.7395
HG	lbs	597	272	1536	1536
MP	lbs-in	808	252	6463	6463
ta	in	0.1064	0.0766	0.1744	0.1744
tb	in	0.1313	0.0522	0.3390	0.3390
tc	in	0.1313	0.0766	0.3390	0.3390
A'b	in^2	0.1137	0.0525	0.7502	0.7502
t1	in	0.1171	0.0766	0.1918	0.1918
t2	in	0.1444	0.0766	0.3729	0.3729
Ab	in^2	0.1193	0.0525	0.7877	0.7877
t1d	in	0.787	0.512	1.393	0.79
t2d	in	N/A	0.512	N/A	N/A
Abd	in^2	1.088	0.271	1.085	1.085

The most nomenclatures of the Table can be found from section 5.1 of this note, also from the appendix 2 and appendix Y of ASME VIII, Div.1, except:

t_1, Estimated calculated flange 1 thickness
t_2, Estimated calculated flange 2 thickness
A_b, Estimated calculated root diameter area of the bolts.
t_{1d}, Designated flange 1 thickness
t_{2d}, Dersignated flange 2 thickness
A_{bd}, Designated root diameter area of the bolts.

Per table 5.3, it is found that all $t_1 > t_{1d}$; $t_2 > t_{2d}$ and $A_b > A_{bd}$ for all respective flanges, so the flange design are satisfactory.

Per figure UW-13.2, section UW-13, appendix 2 of ASME VIII, Div. 1 and drawing ME-439317, it is found that all flange welds are designed per the related codes.

6. Finite Element Analysis (FEA) For the Vessel Weldment, Bottom Supports, and Top Lifting Fixtures

Reference drawing: ME-439317-1 (2)
Figure: i.1.

Per drawing ME-439317, it is found that there are 4 lifting points which are symmetrical in both in xy and yz planes, it is also found that the 4 bottom supports only mirrored about y axis rotation (assuming that the center of the vessel is the rotational point).

It is found that:

W_{ws} = 2,264 lbs. the weight of the vessel shell weldment
$W_{other} \approx$ 3,005 lbs.,
The weight of coldmass shields, support posts, support tubes weldment, other attachments and mice. accessories.
$W_{tot} \approx W_{ws} + W_{other}$ = 5,269 lbs

Figures 6.1 & 6.2 are the calculated stresses from the FEA simulating model, which is based on the 3.9 MHZ vessel weldment with the following boundary conditions:

W_{ws} = 2,264 lbs applied to the model based the vessel physical properties;
$W_{other} \approx$ 3,005 lbs external additional force applied to the model through the 2 top surfaces of coldmass port;
All (4) bottom saddle supports are assumed as simple support.

Figure 6.1, Stress Values in X, Y & Z directions

```
                                                                          Page 1

I-DEAS 11 NX Series m2:    Simulation                       30-Aug-05  09:34:06
C:\FERMI_IDM_PPD92543\ECC_HARMONIC_VESSEL_39MHZ_081805.mf1
stresses for  vessel w/3000# additional load, 4 bottom supt

Group ID           : None
Result Set         : 3 -  B.C. 1,STRESS_3,LOAD SET 1
Report Type        : Contour                        Units          : IN
Result Type        : STRESS
Frame of Reference: Part                            Data Component: Y-Component


        Stress-XX  Stress-XY  Stress-YY  Stress-XZ  Stress-YZ  Stress-ZZ


              96394       6352       9108      22276      34500     122207
Maximum   1.933E+03  1.104E+03  2.583E+03  7.107E+02  1.293E+03  3.269E+03

               2532       2532      21997      96344      33958     121222
Minimum  -3.077E+03 -1.586E+03 -3.044E+03 -6.705E+02 -1.357E+03 -3.119E+03


Average   2.317E+01 -2.425E-01 -2.731E+01  2.704E+00 -3.065E-02  2.336E+00
```

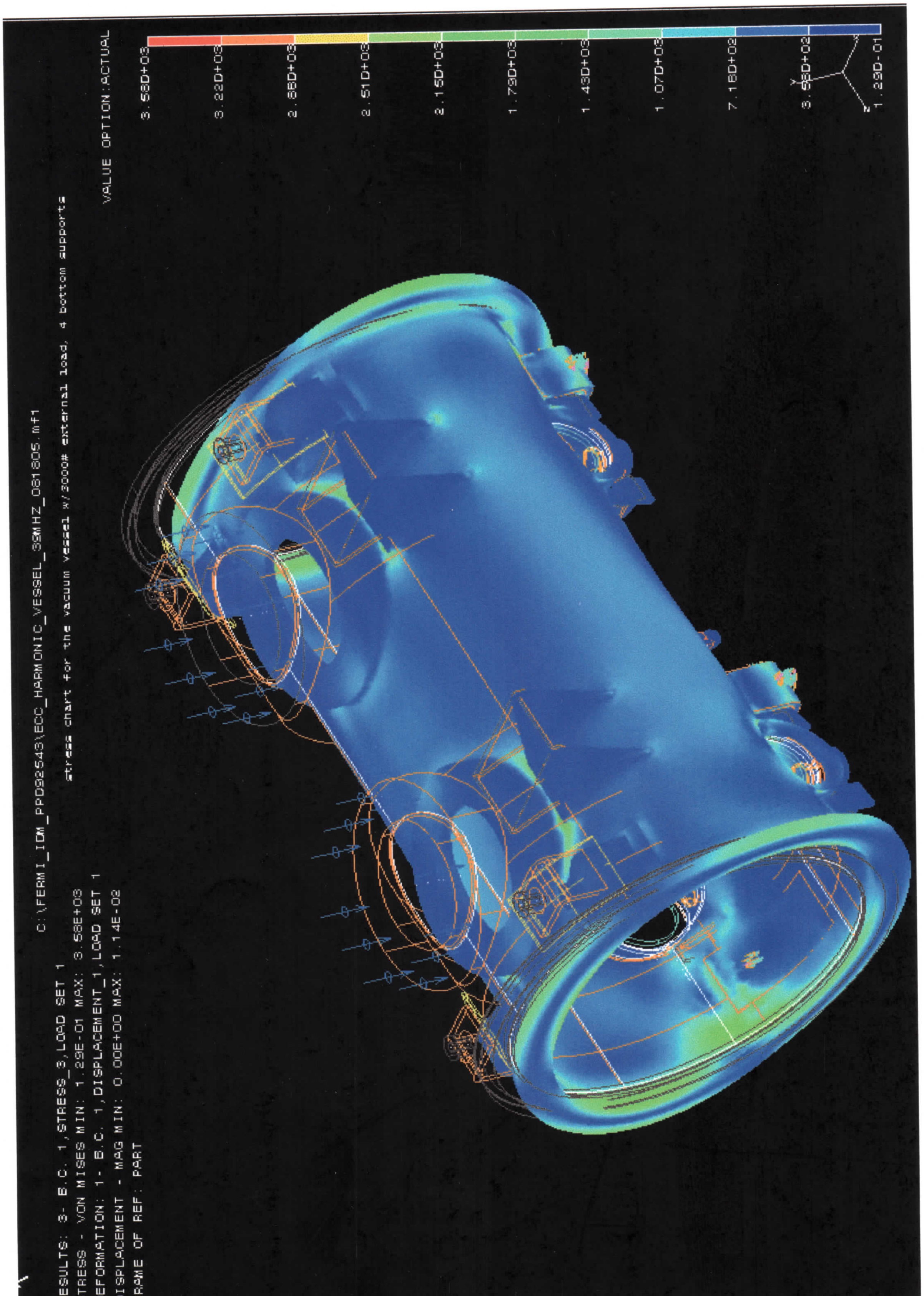

Figure 6.2, The Von Mises Stress & the Deform.Display of the Vessel w/4 Bottom Supts.

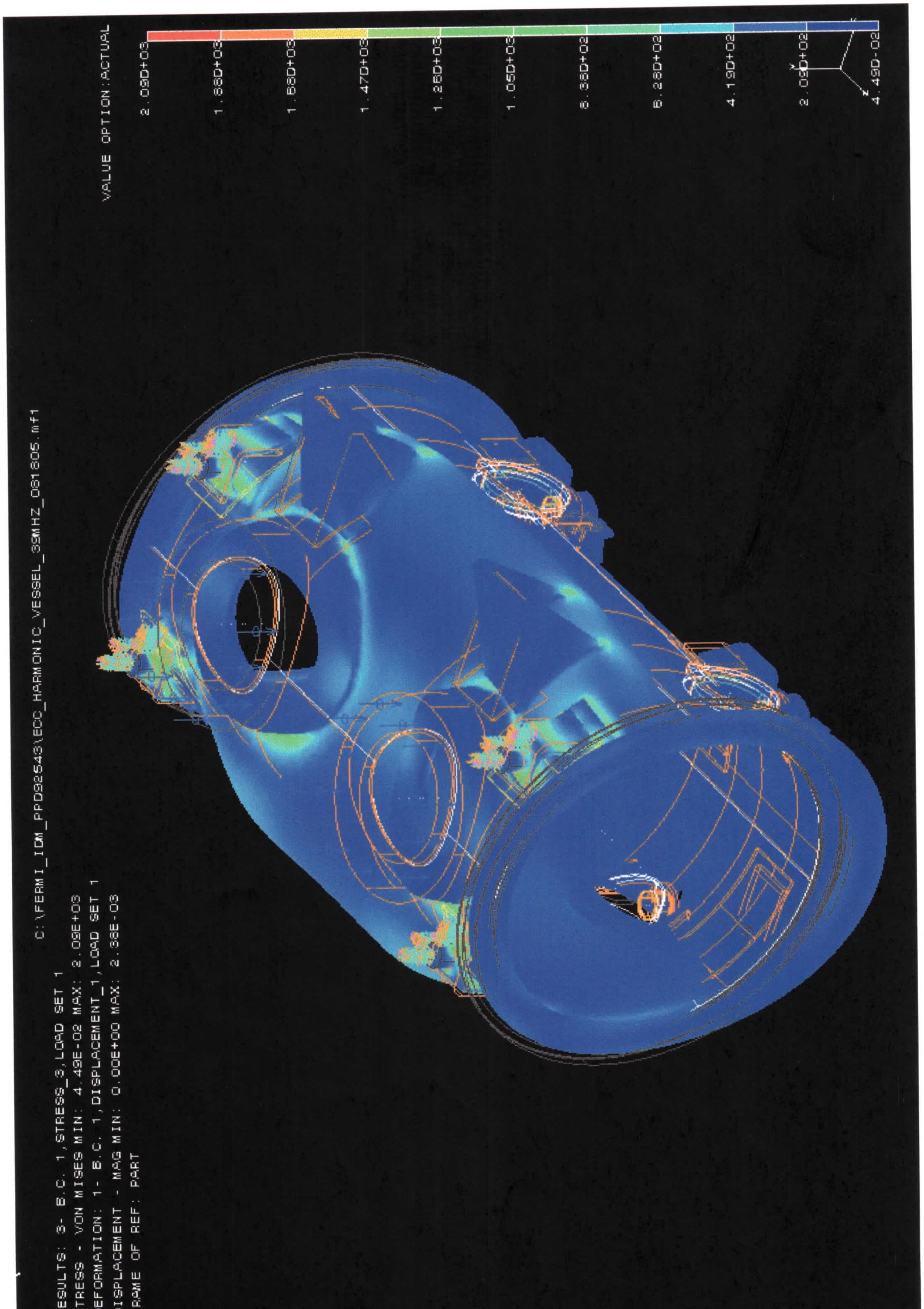

Figure 6.3, The Von Mises Stress & the Deform.Display of the Vessel w/4 Top Lift Pts.

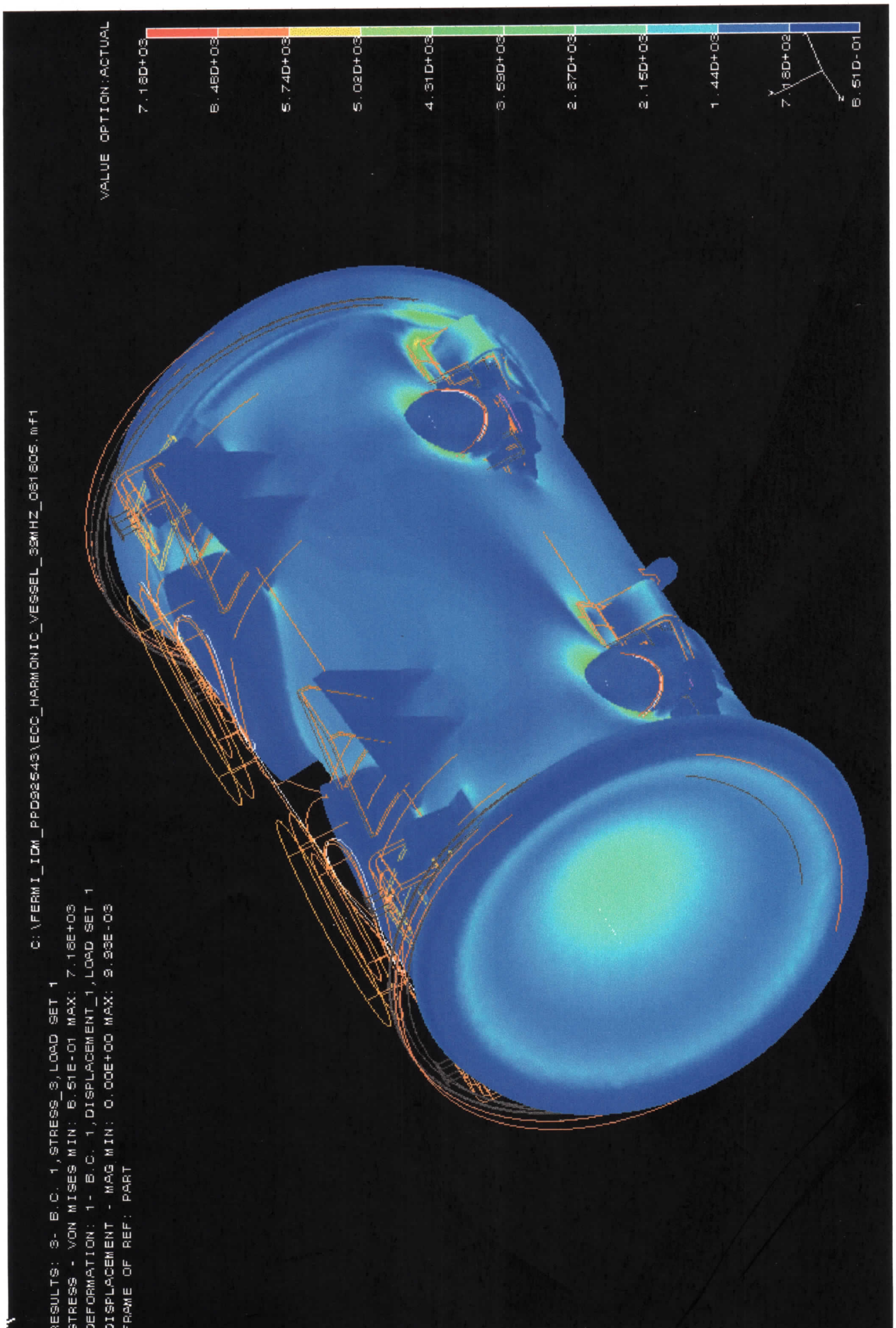

Figure 6.4, The Von Mises Stress & the Deform.Display of the Vessel w/vacuum Cond.

It is found that the maximum Von Mises stress is about $f_{vm} = 3.58$ ksi < F_a=15 ksi, The higher stress areas are: bottom pad base support areas and port opening area, however, all are far below the allowable stress F_a.

Figure 6.3 is the Von Mises stress and deformation display of the vessel weldment model, which is simulating the vessel under the same boundary condition except to lift the vessel from 4 top pick points (instead of supporting by 4 bottom locations). It is found that the maximum Vom Mises stress is about 2.08 ksi, and the max. displacement is about 0.0024".

The simulating results of the vessel under the vacuum working environment can be found from Figure 6.4, the following are the lists of boundary condition:

1. All open flanges and ports are connected with blind flanges with the matching thickness.
2. All outside surfaces of the sealed vessel experienced pressure of 14.7 psi. (Assuming all internal surfaces under pressure of zero)
3. W_{ws} = 4,429 lbs. the weight of the vessel shell weldment and the blind flanges.
   $W_{other} \approx 3{,}000$ lbs.,
   The weight of coldmass shields, support posts, support tubes weldment, other attachments and mice. accessories.
   $W_{tot} \approx W_{ws} + W_{other} = 7{,}429$ lbs
3. All 4 bottom saddle supports are assumed as simple support.

It is found from Figure 6.4 that the maximum Von Mises is about $f_{vm} = 7.18$ ksi, and the most stress area is located around the top corner area of the pad base plate of the bottom support which is next to the upstream or downstream flange. The transition areas from the end flange to the vessel shell, and the main coupler port opening areas are also the relatively higher stress area, however, all calculated stress values are less than the allowable stress F_a under the current boundary condition. It is also found that the max. displacement (~0.01") is around the center of the end blind flange, that is due to the vacuum pressure. All the results from FEA data confirm that the design of the bottom supports, top lifting fixtures and the vessel weldment assembly are satisfactory per the applicable codes.

7. Appendix

7.1 The Lists of the Design Drawings of 3.9 MHz, 3rd harmonic Vacuum Vessel (Fermilab)

1.	Vacuum Vessel Weldment Assy.	ME – 439317 – 1 (2)
2.	Vessel Shell	ME – 439262
3.	Port Fixed Coldmass	MD – 439251
4.	Port Sliding Coldmass	MD – 439252
5.	Pick Point Weldment	MC – 439257
6.	Pad Weld Upper	MC – 439242
7.	Vessel Support Weldment	MC – 439254
8.	Pad Base Support	MC – 439241
9.	Flange MC Support	MD – 439243
10.	Flange Vessel Electronics	MD – 439246
11.	Bracket, Weldment MC LH	MC - 439238
12.	Bracket, Weldment MC LH	MC – 439239
13.	Flange, Cryostat End	MD - 439240
14.	Rib, Stiffening	MC – 439253
15.	Gusset, Outrigger	MC – 439247
16.	Plate, Outrigger	MC – 439248